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ABSTRACT

With growing global warming concerns about current HFC refrigerants, a search for more environmentally-friendly fluids has begun. Potential alternatives to R134a should have significantly lower global warming potential (GWP), operate at similar system pressures, and maintain other chemical and physical property advantages of R134a.

This study investigated four potential alternatives that have been identified by the Air-Conditioning, Heating, & Refrigeration Institute (AHRI) in its Low-GWP Alternative Refrigerants Evaluation Program. These refrigerants were R1234yf, R1234ze, N-13a (HDR-17), and N-13b (HDR-15). Each of these refrigerants was experimentally examined and compared to R134a in a two-phase pumped loop cooling system using a specifically designed test stand. This test stand included a liquid pump, an evaporator with a heating component, and an air-cooled condenser. For each test, refrigerant volume flow rate was varied to achieve the desired exit quality at a specified heat load while maintaining a constant amount of subcooling. The heat loads were varied from 500 W to 1000 W, while the exit quality ranged from 30% to 80%.

R1234yf required substantial volume flow rate increases to achieve similar cooling effects as R134a; however, R1234ze, N-13a, and N-13b appeared to be much more viable solutions, due to smaller increases in volume flow rate required. A number of variables were examined to determine the thermal performance of each refrigerant, including heat sink surface temperatures, heat sink inlet and outlet refrigerant temperatures, and heat transfer coefficients.

At 30% exit quality and a heat load of 500 W, R1234yf, R1234ze, N-13a, and N-13b required 33.13%, 13.14%, 12.54%, and 9.35% higher volume flow rates, respectively, compared with R134a. At 80% exit quality and 1000 W heat load, these differences were 36.65%, 13.61%, 15.12%, and 8.67%, respectively. Increases in required flow rates also resulted in higher pump power consumptions and increased system pressure drops compared with R134a.

1. INTRODUCTION

1.1 Background

In recent years, the protection and preservation of the environment has been a growing issue throughout the world. The refrigeration industry has been involved in this, due to the adverse effects that many commercial refrigerants have on the environment. A number of refrigerants have already been phased out of production, and more could follow.

One of the major agreements dealing with the regulations of refrigerants is the Montreal Protocol. This international treaty was designed to phase out substances contributing to the depletion of the ozone layer. The treaty has phaseout timeframes for various classes of ozone-depleting substances.

The production of the majority of substances with ozone-depleting potentials characterized as high was scheduled to have a calculated level of production of zero by January 2011. These substances include Chlorofluorocarbons (CFCs) and Halons.

In order to accomplish this phaseout in a timely manner, new substances were created and implemented in systems as substitutes. Many of these new substances are Hydrochlorofluorocarbons (HCFCs). Although HCFCs contain less chlorine than CFCs, making them less damaging to the ozone layer, they still contribute to ozone depletion. Therefore, their phaseout was also included in the Montreal Protocol. The Montreal Protocol states that the calculated level of production for HCFCs should not exceed ten percent by January 2016, and the level of production should be zero by January 2031. (EPA, Ozone Depleting Substances)

In addition to their damaging effects to the ozone layer, HCFCs also have high global warming potentials. The Kyoto Protocol was enacted in order to reduce global warming. One method of reducing global warming is to minimize the amount of greenhouse gas released into the atmosphere. Greenhouse gases include Carbon Dioxide, Methane, Nitrous Oxide, Hydrofluorocarbons (HFCs), Perfluorocarbons, and Sulphur Hexafluoride. A widely used refrigerant, R134a, is included in the HFC class. Although the Kyoto Protocol has not yet put a date on it, the complete phaseout of R134a appears inevitable. Therefore, much research is being done and new refrigerants are being evaluated to find a replacement for R134a under the AHRI Low-GWP Alternative Refrigerants Evaluation Program (AHRI, 2012).

A number of refrigerants are being considered for the replacement of R134a (Wang et al, 2012). Among these are R1234yf, R1234ze, N-13a, and N-13b, which operate at similar pressures and temperatures. While R1234yf and R1234ze are pure fluids, N-13a and N-13b are blends. N-13a is a mixture of R1234yf, R1234ze, and R134a, and N-13b is a mixture of R1234ze and R134a. Both have higher GWPs than R1234yf and R1234ze, but lower than R134a. (Honeywell, 2013). Important properties of R134a and 4 other candidates are shown in Table 1.

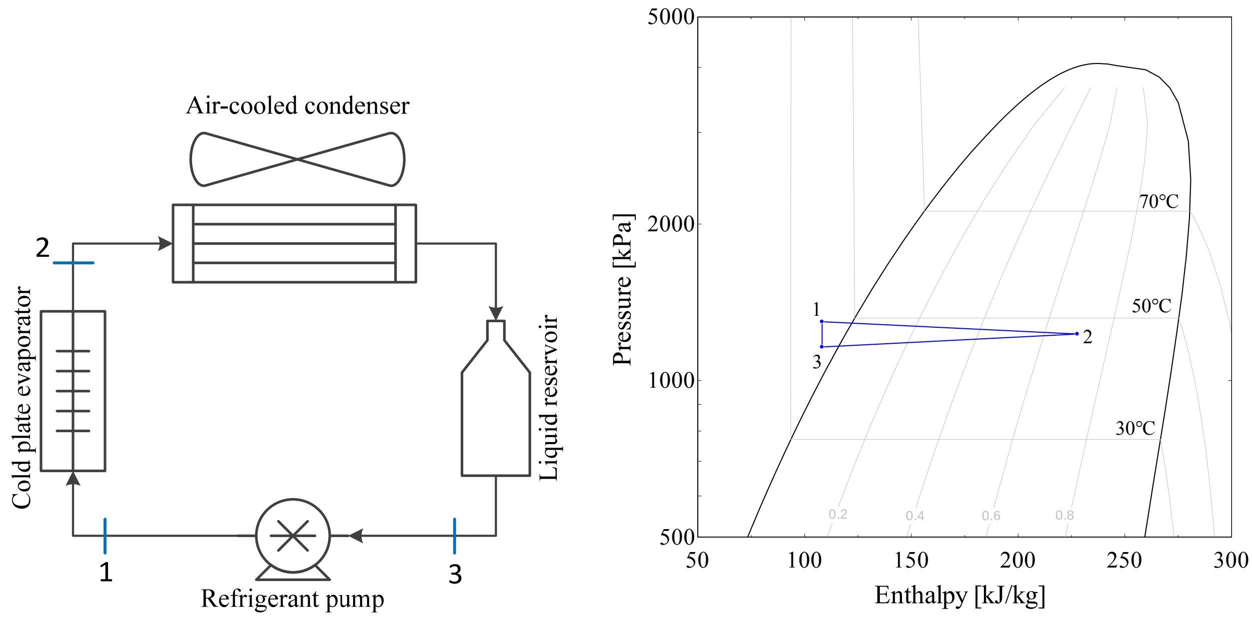
Table 1: Properties of refrigerants*

	Units	R134a	R1234yf	R1234ze	N-13a	N-13b
Chemical name /Composition		1,1,1,2-Tetrafluoro ethane	2,3,3,3-Tetrafluoro propene	1,3,3,3-Tetrafluoro propene	R134a: 42# R1234ze: 40# R1234yf: 18#	R134a: 42# R1234ze: 58#
Enthalpy of vaporization	kJ/kg	182.28	149.29	170.50	168.71	173.51
Liquid density	kg/m^3	1225.3	1109.9	1179.2	1177.5	1192.5
Vapor density	kg/m^3	27.8	32.8	22.6	27.9	25.6
Liquid specific heat	kJ/kg-K	1.405	1.369	1.366	1.386	1.384
Liquid enthalpy	kJ/kg	227.47	226.60	226.74	229.61	228.81
Vapor enthalpy	kJ/kg	409.75	375.89	397.23	398.32	402.32
GWP (100 Years)		1410	4	6	604	604

* - All properties are calculated at 20 °C.
# - By % mass

1.2 Parker Two-phase Liquid Cooling

R-134a is currently being used in Parker Hannifin's Vaporizable Dielectric Fluid (VDF) cooling systems. In a typical VDF cooling system, schematically described in Figure 1, a pump is used to deliver subcooled liquid refrigerant to a heat sink that is in contact with the heat source. While in the heat sink, the refrigerant absorbs the heat from the source isothermally; that is, the refrigerant remains at a constant temperature. During this heat transfer, the refrigerant becomes a liquid-vapor mixture. The refrigerant then moves through a condenser to reject heat from the system. Liquid refrigerant then accumulates in a liquid receiver, until it is pumped back through the system.



(a) (b)

Figure 1: Parker two-phase liquid cooling system shown (a) as a schematic and (b) on a P-h diagram using refrigerant R134a.

2. EXPERIMENTAL SETUP

The primary components of the test stand were: a pump, filter drier, flow meter, heat sink, air-cooled condenser, and an accumulator. A schematic of the test stand with numbered thermocouples and pressure transducers is shown in Figure 2. Subcooled liquid refrigerant was delivered to the heat sink by a pump. Resistive heaters were used for heat generation. The refrigerant liquid-vapor mixture exiting the heat sink was sent to an air-cooled condenser, where heat was rejected to the environment. The refrigerant entering the accumulator was slightly subcooled. The heat sink's cavity was approximately 98.43 mm L x 31.75 mm W x 3.05 mm H (3.875" x 1.250" x 0.120"). The cavity was filled with a fin pack which had 88 total fins at a density of 1.97 fins per mm (50 fins per inch) with a height of 3.05 mm (0.120").

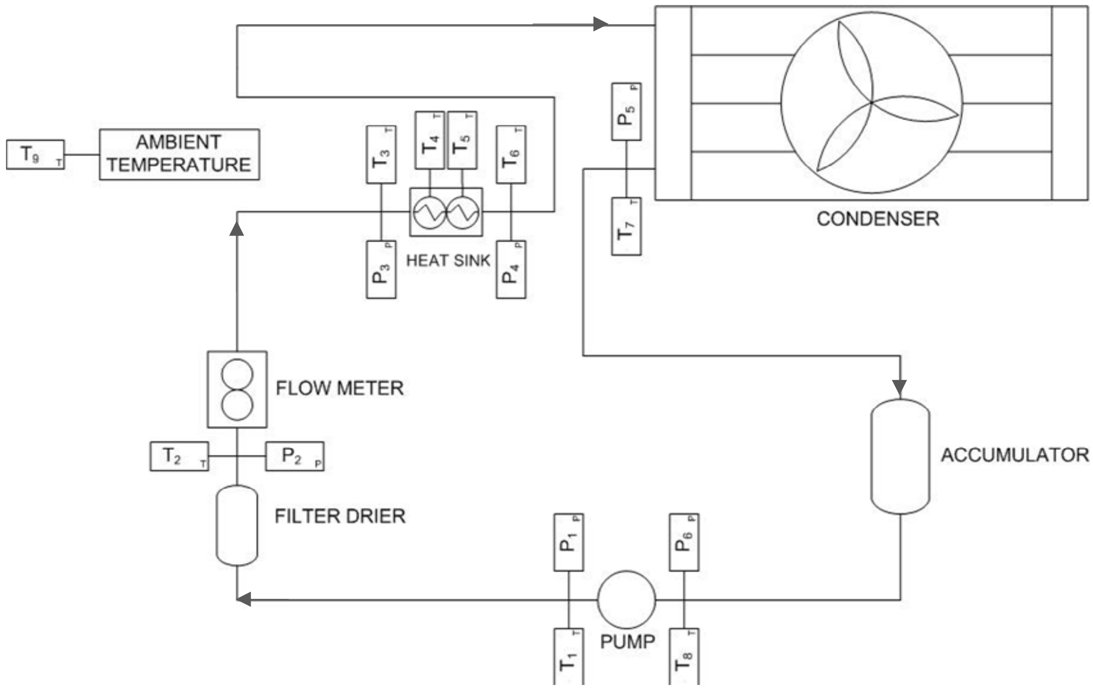


Figure 2: Schematic of test stand with numbered thermocouple and pressure transducer locations.

3. MEASUREMENT AND DATA REDUCTION

3.1 Instrumentation

There were 9 thermocouples and 6 pressure transducers on the test stand. In-stream thermocouple probes were used for temperature measurements T_1, T_2, T_3, T_6, T_7, and T_8. Thermocouples T_4 and T_5 were coated with thermal paste and inserted into holes that were drilled into the heat sink lid.

T_1 and P_1 represent the pump outlet temperature and pressure readings of the refrigerant. T_2 and P_2 represent the flow meter inlet temperature and pressure readings. T_3 and P_3 represent the heat sink inlet temperature and pressure readings of the refrigerant. T_4 and T_5 represent the heat sink surface temperatures. T_6 and P_4 represent the heat sink outlet temperature and pressure readings of the refrigerant. T_7 and P_5 represent the condenser outlet temperature and pressure readings of the refrigerant. T_8 and P_6 represent the pump inlet temperature and pressure readings of the refrigerant. The refrigerant volume flow rate was measured using a paddlewheel flow meter.

LabVIEW, coupled with National Instruments cDAQ modules, was used for recording measurements. This system was also used to send control voltages to the heating elements, pump, and condenser fan. In order to verify the accuracy of measurements obtained from the data acquisition system, a number of calibrations and verifications were performed. The pressure transducers used in the test stand were calibrated. The thermocouples and flow meter were verified.

3.2 Tests

A total of 54 tests were conducted for each of the five refrigerants. The heat load on the heat sink was varied from 500 W to 1000 W by 50 W increments. The vapor quality of refrigerant at the heat sink outlet was varied from 30% to 80% by 10% increments. In order to achieve the desired exit quality, refrigerant flow rate was varied by adjusting the pump control voltage. Subcooling of 2°C was maintained at the heat sink inlet for all tests by adjusting the control voltage to the condenser fan.

3.3 Data Reduction

Refrigerant outlet and inlet qualities at the heat sink evaporator are calculated using Equations (1) and (2) respectively.

$$x_{out} = x_{in} + \frac{Q}{\dot{m} \cdot h_{fg}} \tag{1}$$

$$x_{in} = \frac{h_{in} - h_f}{h_{fg}} \tag{2}$$

$$\dot{m} = \frac{\dot{V}}{v_{in}} \tag{3}$$

where:

x_{out} = Quality of refrigerant at heat sink outlet
x_{in} = Quality of refrigerant at heat sink inlet
h_{in} = Specific enthalpy of refrigerant at heat sink inlet
h_f = Specific saturated liquid enthalpy of refrigerant at pressure P_2
h_{fg} = Specific enthalpy of vaporization; h_f calculated from P_2 and h_g calculated from P_3
Q = Heat load supplied by heaters [W]
$\dot{m}$ = Mass flow rate of refrigerant [kg/s]
$\dot{V}$ = Volume flow rate of refrigerant
v_{in} = Specific volume of refrigerant, calculated using temperature T_2 and pressure P_2

To determine the thermal resistance, the heat sink outlet fluid temperature was subtracted from the average heat sink temperature, which was then divided by the heat load, as seen in Equation (4).

$$R_{\mathrm{th}} = \frac{(T_4 + T_5)/2 - T_6}{Q} \tag{4}$$

where:

R_{th} = Thermal resistance [K/W]
T_4 = Heat sink thermocouple reading [°C]
T_5 = Heat sink thermocouple reading [°C]
T_6 = Refrigerant temperature exiting heat sink [°C]

The heat transfer coefficient was calculated by dividing the heat load by the product of the heater area and the log mean temperature difference, as seen in Equation (5).

$$h_{meas} = \frac{Q}{A_{surf,heat} \cdot \Delta T_{LM}} \tag{5}$$

where:

$A_{surf,heat}$ = Surface area of applied heat load [m^2]
ΔT_{LM} = Log Mean Temperature Difference

To calculate the log mean temperature difference, ΔT_{LM}, Equations (6), (7) and (8) were used.

$$\Delta T_{LM} = \frac{\Delta T_2 - \Delta T_1}{ln\left(\frac{\Delta T_2}{\Delta T_1}\right)} \tag{6}$$

$$\Delta T_2 = T_5 - T_6 \tag{7}$$

$$\Delta T_1 = T_4 - T_3 \tag{8}$$

where:

T_3 = Refrigerant temperature at heat sink inlet [°C]
T_4 = Heat sink thermocouple reading [°C]
T_5 = Heat sink thermocouple reading [°C]
T_6 = Refrigerant temperature at heat sink outlet [°C]

4. RESULTS

Tests were conducted with the aforementioned refrigerants at various refrigerant exit qualities and heat loads, as described in the previous section. Figure 3 shows the refrigerant volume flow rate versus heat load at a constant exit quality of 70% for each refrigerant. The volume flow rate linearly increased with increases in the heat load. Due to its higher latent heat of vaporization and density, R134a required the lowest volume flow rate, while R1234yf required the highest.

The volume flow rate versus exit quality was plotted for a heat load of 500 W, as shown in Figure 4. Increasing exit quality led to a drop in flow rate. R134a required the lowest volume flow rate at any given exit quality of all refrigerants tested, indicating better thermal performance than the other refrigerants. Conversely, R1234yf required the highest refrigerant volume flow rate.

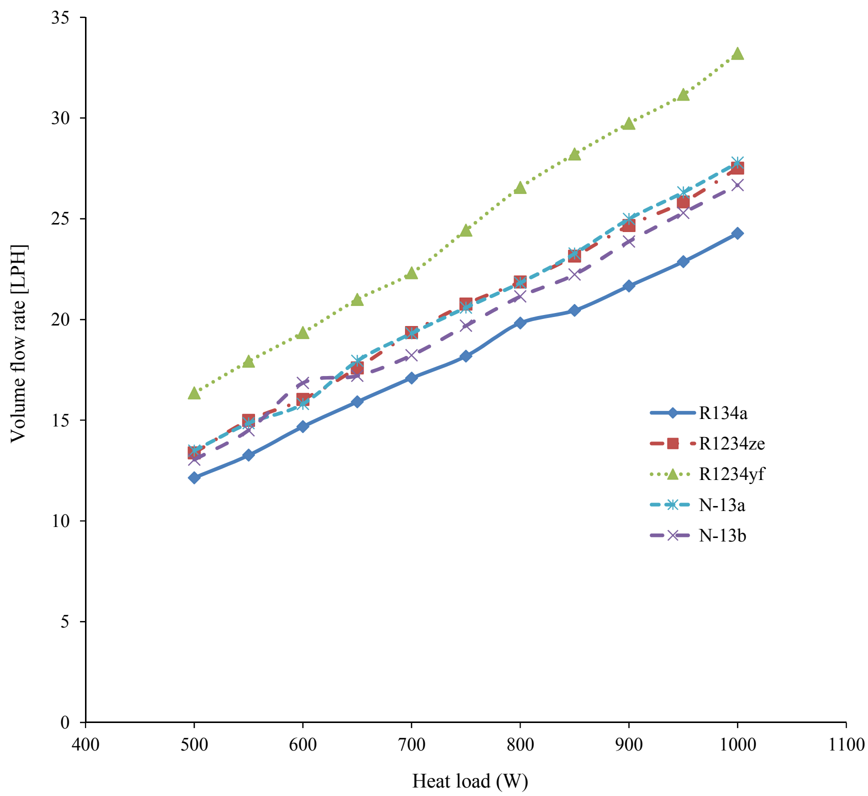


Figure 3: Variation of refrigerant volume flow rate as a function of heat load at a constant exit quality of 70%.

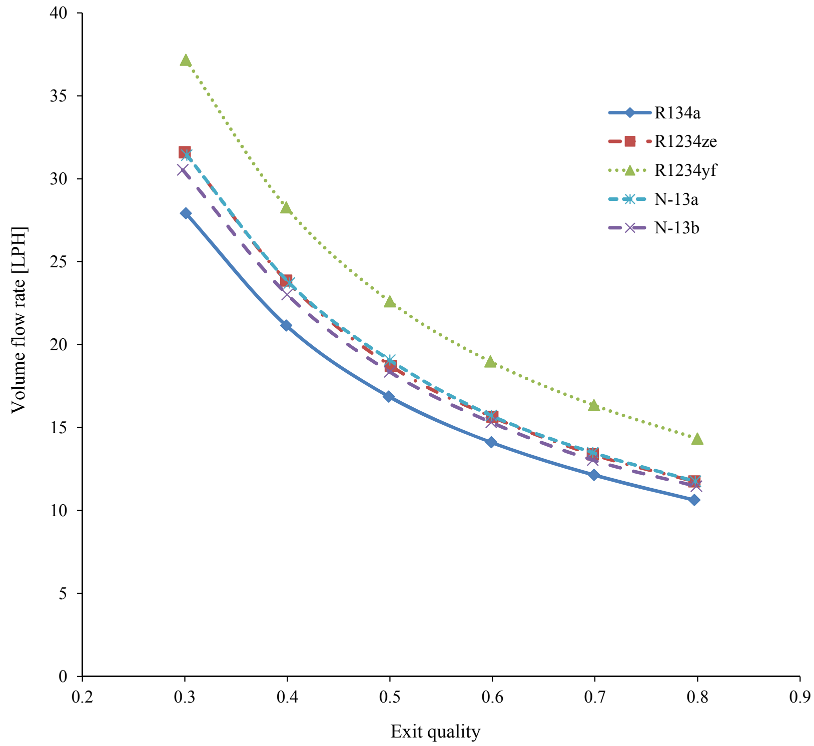


Figure 4: Variation of refrigerant volume flow rate as a function of exit quality at a heat load of 500 W.

Figure 5 compares the volume flow rates of each of the alternate refrigerants tested to those of R134a. For a heat load of 500 W, R1234yf required approximately 34% higher flow rates compared with R134a at all exit qualities, which can be partially attributed to its low liquid density. R1234ze and N-13a required approximately 11% higher refrigerant flow rates compared with R134a. N-13b required approximately 8% higher flow rates compared with R134a, making it the best performer based on volume flow rate, which can be partially attributed to its high liquid density.

Figure 6 shows measured pressure differential (ΔP) across the pump for all refrigerants as a function of exit quality for a constant mass flow rate of 0.007 kg/s. Generally, the system pressure drop increased linearly with the exit quality. While the pressure drop for R134a increased from 12.6 kPa to 14.8 kPa for a corresponding increase in the exit quality from 0.3 to 0.8, ΔP values for all alternate refrigerants were found to be significantly higher. On average, R1234ze, R1234yf, N-13a and N-13b showed 46%, 39%, 40% and 42% higher two-phase pressure drops, respectively, than R134a.

Figure 7 shows the calculated average heat transfer coefficient for all refrigerants versus exit quality at a mass flow rate of 0.007 kg/s, which equates to approximately 20 L/hr of R134a at normal operating temperature. In general, the heat transfer coefficients increased with increasing exit quality up to 70%. Between 70% and 80% exit quality, an increase in quality led to a drop in the heat transfer coefficient. For all exit qualities, R134a exhibited the highest heat transfer coefficient.

This trend of the heat transfer coefficient first increasing with vapor quality, reaching a maximum and then decreasing with further increase in quality has also been observed by Bertsch et al. (2008). While Bertsch et al. (2008) measured the highest heat transfer coefficient at 50% quality for R34a; they noted that this peak depended on fluid, geometry, and flow conditions.

Figure 8 compares the heat transfer coefficients of each of the alternate refrigerants tested to those of R134a. All of the alternate refrigerants tested were found to have lower heat transfer coefficients than R134a. The heat transfer coefficients of R1234yf were closest to those of R134a. This was in sharp contrast with Figure 5 which showed R1234yf as the worst performing refrigerant in terms of required volume flow rate.

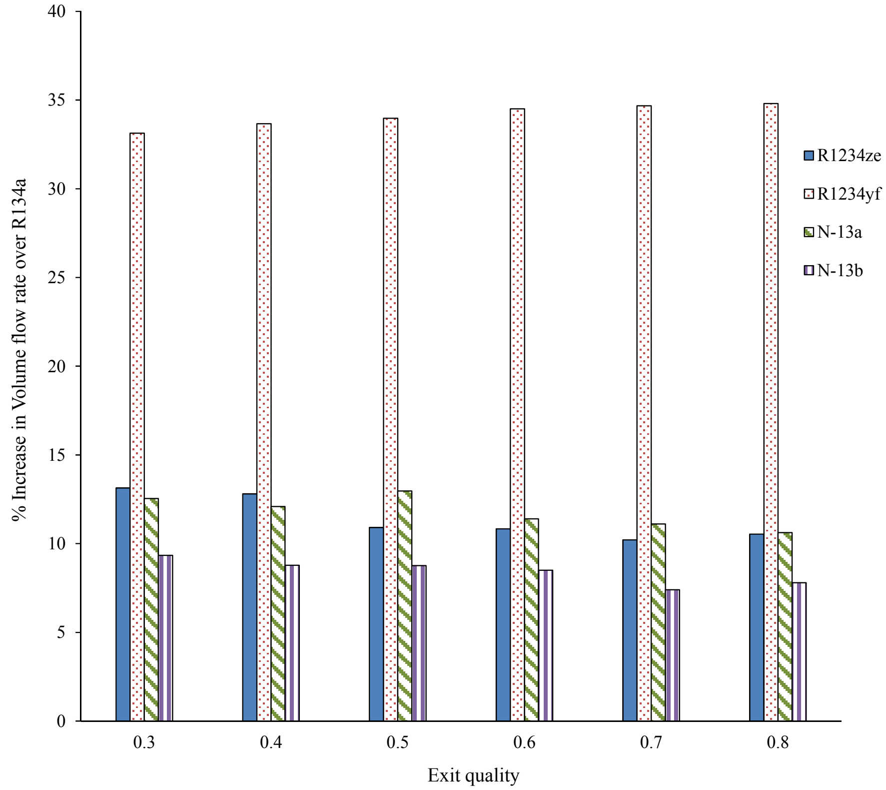


Figure 5: Percentage change in required refrigerant volume flow rates for candidate refrigerants compared to R134a as a function of exit quality at a heat load of 500 W.

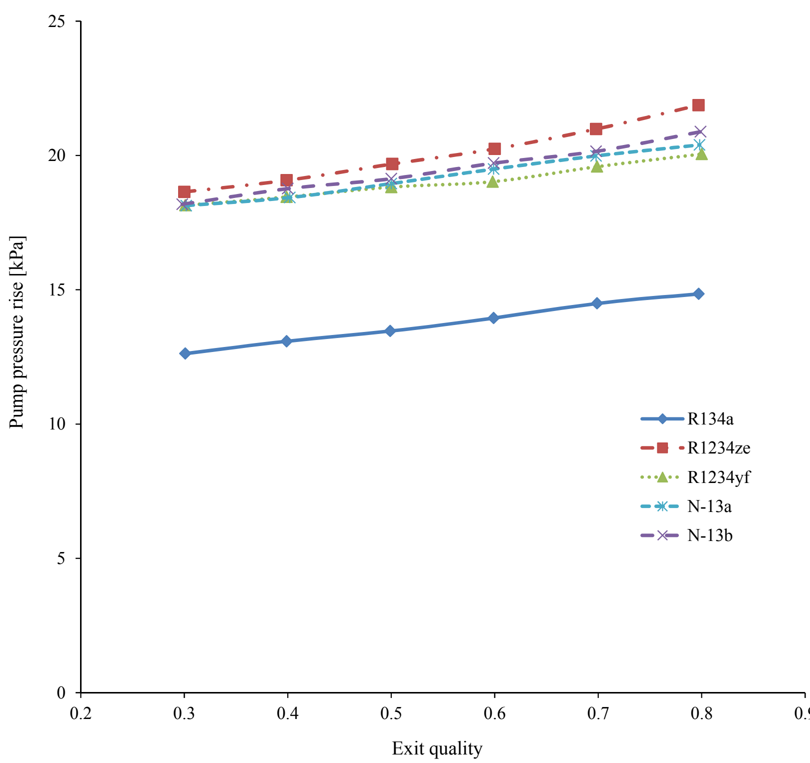


Figure 6: Pressure differential across the pump as a function of exit quality for a constant mass flow rate of 0.007 kg/s.

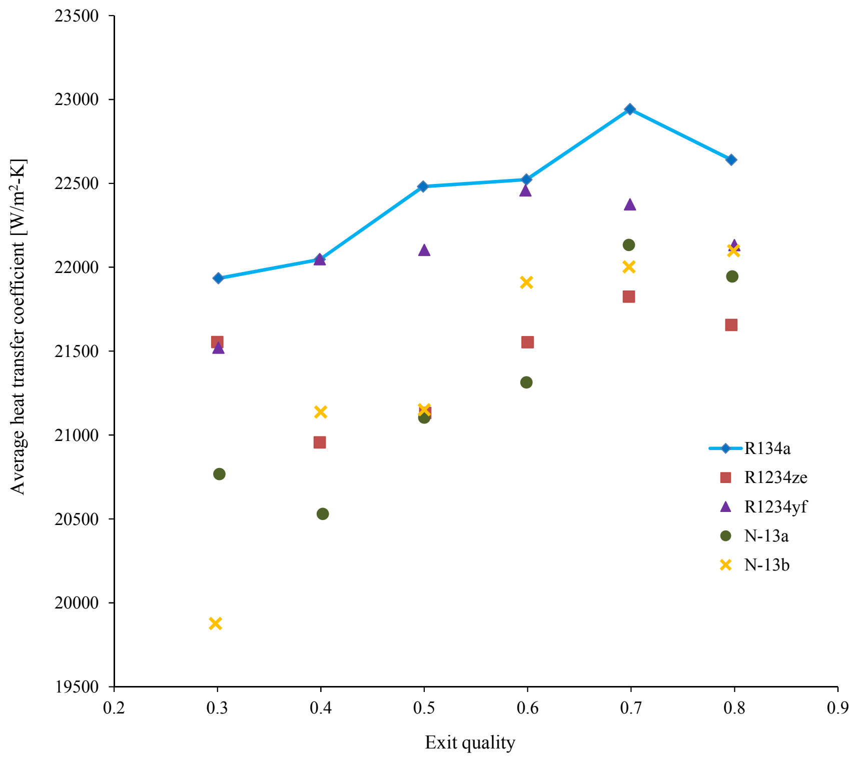


Figure 7: Measured average heat transfer coefficients as a function of exit quality for a constant mass flow rate of 0.007 kg/s.

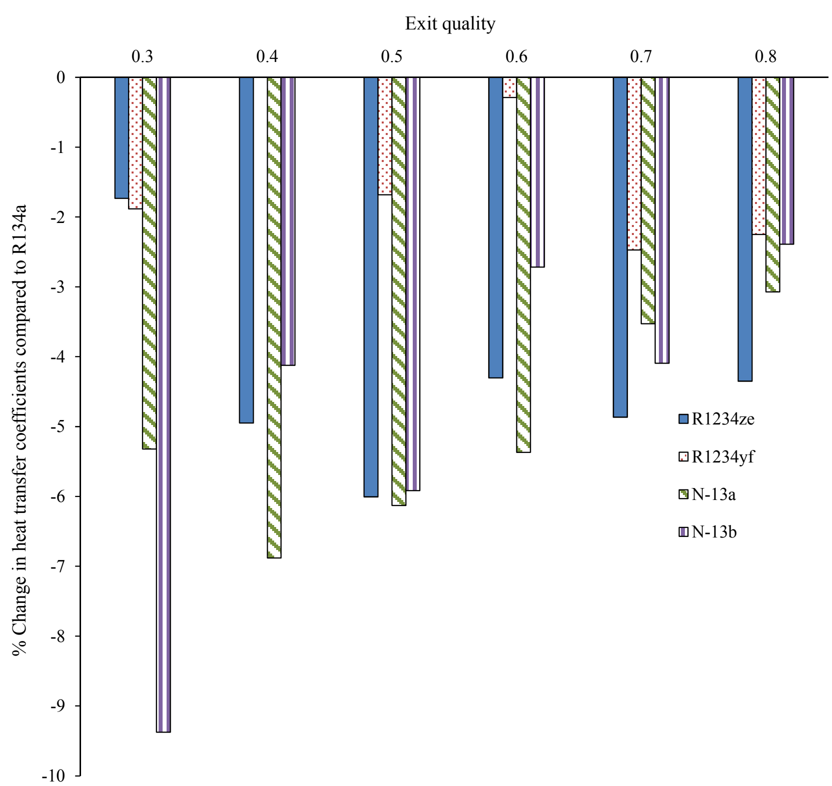


Figure 8: Percentage change in heat transfer coefficients for candidate refrigerants compared to R134a as a function of exit quality at a constant mass flow rate of 0.007 kg/s.

5. CONCLUSIONS

Four low-GWP refrigerants, R1234ze, R1234yf, N-13a and N13b, for the possible replacement of R134a were experimentally tested and benchmarked with R134a for use in a two-phase liquid cooling system. The tests were conducted at various heat loads and refrigerant exit qualities. Refrigerants were compared based on their volumetric flow rates at given heat loads and calculated heat transfer coefficients at a given mass flow rate.

The results indicated that R134a had the best performance in terms of its required volume flow rate for a given heat load as well as its overall heat transfer coefficient and two-phase pressure drop at a given mass flow rate. While the former characteristic is important for pump sizing, the latter can have significant impact on the overall thermal resistance. With an average volume flow rate increase of 8%, N-13b exhibited the best performance in terms of flow rate when compared to R134a. R1234yf closely matched the heat transfer coefficient values of R134a. R1234yf also showed 39% higher two-phase pressure drop which was better than the other candidates.

These findings are listed in Table 2 that also shows important selection criteria and the best suited candidate for each of these criteria.

Table 2: Important system design parameters and suitable refrigerant

Criteria	Importance	Candidate
GWP	Environment	R1234yf and R1234ze
Volume flow rate	Pump sizing	N-13b
Pressure drop	Pump power consumption	R1234yf
Heat transfer coefficient	Heat sink thermal resistance	R1234yf

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